N74-10240

Paper No. 15

THERMAL SCALE MODELING OF A SPACECRAFT RADIATOR WITH COUPLED FORCED CONVECTION-CONDUCTION-RADIATION HEAT TRANSFER

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ABSTRACT

A thermal model of a spacecraft radiator has been designed and tested at the National Aeronautics and Space Administration's Mississippi Test Facility. The unique feature of this model is that all three modes of heat transfer--forced convection, conduction, and radiation--are utilized simultaneously under steady-state conditions. A fluid is forced under pressure using similar flow conditions through both prototype and model which are suspended within a cryogenic vacuum chamber. Heat is transferred from the fluid to the tube's inside wall, which then conducts the energy to its outer surface where it is radiated to the surrounding shell that is maintained at a cryogenic temperature with liquid nitrogen. A high vacuum chamber at the facility is used to house the experiment. Both prototypes and models take the form of long tubes with thermocouples welded to the exterior surface to determine the effectiveness of the modeling criteria. Special precautions have been taken to isolate thermally the specimen and to establish a hydrodynamic boundary layer before specimen entry. The wall thickness of the models has been sized to permit both temperature and material preservation. The effects of physical size and fluid flow parameters on the modeling criteria for both low and high thermal conductivity materials are presented.

BACKGROUND

Measurement of temperature plays an important role in design and development of objects or systems which are exposed to hostile environments. An environment such as outer space with its high vacuum and low temperature may be simulated, for test purposes, with a cryogenic vacuum chamber. These space chambers are limited in size and may require extensive supporting facilities. For these and other reasons, use of scale models for test purposes has become expedient and sometimes necessary.

A thermal model may be defined as a model, different and usually smaller in size than its prototype, that will accurately predict the thermal behavior of its prototype under suitable conditions.

A radiator will be used to control the environment within the spacecraft during such extended missions as that proposed for Skylab. These radiators will transfer heat energy from a fluid which has been circulated through the living quarters and electronic equipment. This energy will then be radiated in a controlled manner to deep space so that a suitable environment can be maintained within the spacecraft. Space radiators will be used on Skylab's Apollo Telescope Mount and the 14-foot-diameter Space Station Module. Future radiators such as those for a space Shuttle Vehicle may be extensive in size and could require elaborate test facilities.

In order to model thermally a given object or system accurately, the scale factors or ratios of model-to-prototype parameters must be determined. Thus, one may observe the behavior of the parameter of interest--for example, temperature--on the model; and by application of the scale factor, he may determine what the parameter would be in a corresponding location on the prototype or full-scale specimen.

The period of time during which the parameter of interest is observed is important for analysis purposes. Equilibrium conditions which may occur during long periods of space travel may be successfully modeled as steady-state conditions. Time periods during which parameters may vary, such as launch, midcourse correction, and reentry, involve transient conditions.

Thermal energy or heat is transferred due to a difference in temperature and depends upon the nature of the surrounding medium. Heat may be transferred by conduction through solids or fluids due to direct contact of mass. Convective heat transfer occurs between a fluid and a surface and depends upon the motion; e.g., velocity of the fluid relative to the surface. Free or natural convection involves fluid flow due to a density gradient whereas forced convection occurs when the fluid is forced to flow because of a difference in pressure. Radiation heat transfer or infrared electromagnetic radiation does not require an intermediary medium and becomes increasingly important with large temperature differences.

Previous work in thermal modeling has involved mainly steady-state conduction and conduction-radiation coupled systems such as may be found in the walls of an unmanned spacecraft during a long interplanetary voyage. Some investigations into transient modeling of these systems have also been accomplished. Convection-conduction-radiation coupled systems as encountered in fluid systems and manned spacecraft have only recently and partly been investigated. Steady state and transient analyses on a system of concentric cylinders with free convection within an annulus was completed in 1969; but, prior to this research, no work had been published on convection-conduction-radiation coupled systems involving forced convection.

The purpose of this research was to investigate the applicability of thermal modeling under steady-state conditions for a single material system involving forced convection from a flowing fluid in a tube, conduction through and down the tube, and radiation to a cryogenic vacuum environment.

Investigations into thermal modeling of spacecraft and their components began less than 10 years ago. Some of the studies included experimental programs, while others were theoretical. Numerical analysis has been frequently used to verify proposed modeling criteria. Most of the studies involved coupled conduction and radiation systems under steady-state and transient conditions. Jones [1] used the similitude method to reduce a set of simultaneous, first-order differential equations which described the thermal behavior of spacecraft to a group of 28 ratios that were required to remain constant. These ratios contained six independent sets. Rolling [2] used the similitude approach to develop the modeling criteria for space vehicles. Adkins [3] introduced a method of geometric scaling that allowed thermal modeling while preserving both material and temperature. He utilized similitude to develop modeling criteria for a thin-walled cylinder. Miller [4] investigated the application of thermal modeling to steady-state and transient conduction in cylindrical solid rods for both single and multiple material systems. Maples [5] was evidently the first to investigate thermal modeling with all three modes of heat transfer simultaneously. He analyzed the problem of free or natural convection in the annulus of a concentric cylinder system. The similitude approach was applied to the energy differential equation to obtain the modeling criteria. Both temperature and material preservation were employed, and the diameter was scaled as $D^* = L^{*2}$. Thermal energy was conducted radially from a heater within the inner cylinder through the wall to dry air within the annulus. Following the free convection across the annulus, the heat was conducted through the outer cylinder and radiated to the cryogenic liner surrounding the inside of the vacuum chamber. MacGregor [6] at Boeing analyzed the limitations associated with thermal modeling. An understanding of errors resulting from uncertainties in the thermophysical properties, geometric dimensions, and the test environment was the primary objective of this study. Rolling, Murray, and Marshall [7] at Lockheed also discussed the limitations associated with thermal scale modeling at length. It was concluded that the problems regarding model construction, instrumentation, and materials selection become increasingly difficult at the smaller scale ratios. Temperature preservation was preferred over material preservation, and the use of both techniques simultaneously required geometric distortion of all components which could become difficult in most complex systems. Colvin and Maples [8] outlined the procedures for this experiment in April 1971 and presented preliminary data for one stainless steel model and prototype. Colvin [9] reported the complete results of this experiment in June 1971.

MODELING CRITERIA

Thermal modeling has been divided into two categories: temperature preservation and material preservation. Temperature preservation required that temperatures at analogous locations on the prototype and model be equal. In some cases this may require that a different material be used for the model than for the prototype. Material preservation permits the use of the same material for both prototype and model, but predicts a scaled difference in temperature at analogous locations. Most researchers prefer to maintain thermal similitude between prototype and model, but it would also be desirable to use the same material for both objects. This combination of criteria has been used by Miller [4], Adkins [3], and Maples [5]; and appears to be satisfactory under certain circumstances. The restriction involves the use of a thin-wall approximation which may be acceptable depending upon the object being modeled. For the case of a thin-walled tube or chamber such as a spacecraft wall, this approximation may be used to develop certain modeling criteria.

Further, thermal modeling may be approached in two ways: dimensional analysis or similitude. Dimensional analysis requires knowledge of all parameters associated with the problem, but can lead to useful results. The similitude approach involves the use of the governing differential equations and boundary conditions and offers a distinct advantage to the inexperienced. Either method results in the same set of similarity parameters, but the similitude approach will be used here.

Before deriving the similarity parameters, the constraints imposed upon the problem will be discussed. The first restriction involved the use of homogenous and isotropic materials. The second required that there be perfect geometric similarity between prototype and model. Thirdly, the model and prototype must have the same uniform and constant surface characteristics. This was achieved by coating the surfaces of both the prototype and model with a highly absorptive flat black paint. The fourth requirement was that the radiant heat flux from the simulated environment was approximately zero. This approximation was achieved by using a cryogenic liner cooled to liquid nitrogen temperature to simulate the environment. It was also assumed that all energy radiated from the prototypes and models was absorbed by the cryogenic liner. The fifth restriction was that the properties of the prototype and model were constant and invariable during testing. Use of a small temperature range of approximately 30 F to 80 F insured this approximation. A sixth constraint was that heat transfer by convection and conduction external to the specimen was negligible. This criteria was satisfied by a vacuum environment, the suspension of the test element on nonconducting threads, and its connection to adjacent tubing with insulated fittings.

With these constraints it was decided to test a low and a high thermal conductivity material to verify the modeling criteria for the forced convection-conduction-radiation problem. A fluid at room temperature with a fully developed velocity boundary layer was introduced to a tubular specimen with a large length-to-diameter ratio. Heat was then transferred from the water to the inner surface of the tube by forced convection. This energy was then transferred through the tube to its outer surface and along its length by conduction. Because the specimen tube was thermally insulated from its connecting members and surrounded by a vacuum environment, the only avenue remaining for heat transfer from the outer surface was radiation to the cryogenic liner. The specimen tube was allowed to achieve thermal equilibrium, thereby satisfying the steady-state criteria.

Similarity parameters were derived by Colvin and Maples [8] from the conduction equation for the temperature distribution in a pipe. The results of the modeling criteria may be summarized as

$$D^* = t_w^* = Nu^* = L^{*2} \tag{1}$$

and

$$T^* = \kappa^* = 1 \tag{2}$$

where Nu is the Nusselt number, κ is the tube's thermal conductivity, t_w is the wall thickness of the tubing, L is the length of the tube, and T is the surface temperature of the tube. Equation 2 implies both temperature and materials preservation. The * indicates a scaled quantity of model-to-prototype ratio.

TEST SPECIMENS

In order to verify the modeling criteria, a series of test models were fabricated of both high and low thermal conductivity materials. A 1.0-inch outside diameter (O.D.) type 304 stainless steel tube 48 inches in length was used as the prototype or full-size low thermal conductivity specimen. Three scale models were then fabricated from 0.75-inch O.D., 0.50-inch O.D., and 0.25-inch O.D. type 304 stainless steel tubing. Their scale lengths were 41.568 inches, 33.936 inches, and 24.000 inches, respectively. A 1.0-inch O.D. type 6061 aluminum tube 48 inches in length was used as the prototype high thermal conductivity specimen. Three scale models were also fabricated from 0.75-inch O.D., 0.50-inch O.D., and 0.25-inch O.D. type 6061 aluminum tubing to the same lengths as those of the stainless steel models.

The scale models were fabricated on a lathe by turning down the O.D. of the tube to the desired wall thickness based upon the modeling criteria given earlier and the average wall thickness of the 1.0-inch O.D. prototype. According to the criteria, the wall thickness scales as the diameter. Thus, the wall thickness of the 0.75-inch O.D. model must be 0.75 the wall thickness of the 1.0-inch O.D. prototype. Likewise, wall thickness of the 0.50-inch model must be 0.50 the wall thickness of the 1.0-inch O.D. specimen, and 0.25-inch O.D. model must have a wall thickness that is 0.25 the wall thickness of the 1.0-inch O.D. prototype. The outside diameter of the models was then turned down on a lathe to yield the desired wall thickness. A short lip of material was left to the original O.D. to facilitate machinability and allow connection of the model with common fittings during testing. Because of the selection of nearest largest-size tubing, this lip was usually only 0.010 inch larger than the turned-down dimension and thus can be considered to contribute little, if any, thermal effect to the temperature measurement near the ends of the tubing.

In order to insulate thermally the specimen tube from the tubing before and after itself, nylon Swagelok unions were used as connections at each end. Teflon front ferrules were used in each fitting to achieve better sealing characteristics of the connection. Additionally, to insure a smooth flow within the tube at the leading end, a teflon insert was fitted within the nylon union and its inside diameter was matched to the inside diameter of the respective tubing.

It was then necessary to fabricate an entrance tube of proper length from the same stock as that of the test specimen so a desirable fluid flow profile or velocity boundary layer within the tube could be established before entry into the test specimen. The inside diameter of the entrance tube, front fitting, and test specimen were then the same; thus avoiding any discontinuities that could induce undesirable turbulence or mixing within the flowing fluid. The length of the entrance tube for the establishment of laminar flow is a function of both Reynolds number and tubing size according to the relation

$$\ell = \left(\frac{d}{20}\right) Re$$

where ℓ was the required tube length, d was the tube diameter, and Re was the dimensionless Reynolds number.

The 1.0-inch O.D. tubing had an entrance tube length of 100 inches. The 0.75-inch O.D. entrance tube was 75 inches in length. Similarly, the 0.50-inch O.D. entrance tube was 50 inches long, while the 0.25-inch entrance tube was 25 inches in length. Plug gages were fabricated from brass or nylon rods and used to insure alignment.

It was necessary to attach thermocouples to the exterior surface of the specimen tube in order to determine thermal similarity between prototype and model. Fourteen 30-gage, copper-constantan thermocouples were fabricated and spot welded to each specimen tube at certain locations (Figure 1). Leads to each thermocouple were wrapped circumferentially around the specimen to minimize lead wire measurement error. The tubes were then spray painted with two thin coats of flat black paint (Velvet coating 101-C10 by 3M) to insure uniform and efficient radiative heat transfer. Thermocouple lead wires were then painted with a bright aluminum paint to a distance at least six inches from the tube to reduce lead wire radiation loss and subsequent measurement error.

EXPERIMENTAL APPARATUS

The experimental apparatus can be divided into eight basic sections: the vacuum and cryogenic system, the tubular system, the instrumentation and recording system, the inlet temperature control system, the flow pressurization system, the flow evacuation system, the flow measuring and control system, and the flow collection system. A schematic diagram of the pressurization and flow systems is given in Figure 2.

Prototypes and models were fabricated for test inside a space simulation chamber that provided the necessary low temperature, high vacuum environment for accurate simulation of energy exchange between the tubes and their surroundings. A Murphy-Miller high altitude test chamber is a standard piece of test equipment located at the National Aeronautics and Space Administration's (NASA) Mississippi Test Facility. This chamber was constructed of carbon steel with an interior 48 inches in diameter, 60 inches long, and had a raised shelf four inches above the bottom. The chamber was evacuated through one end, and a full-width door across its opposite end provided easy access to the interior. Instrumentation feedthroughs in the chamber wall permitted direct connection to the 16 thermocouples; fluid feedthroughs introduced liquid nitrogen to the cryogenic liner. This liner was designed to fit within the chamber like a sleeve and simulate the low temperature environment of outer space. The liner shell was constructed of stainless steel with interior dimensions 54 inches long and 38 inches in diameter. The inner wall of the liner was coated with 3M Velvet Coating 101-C10 black paint to insure a surface with high and uniform values of emittance. The outer wall of the liner and the inner wall of the chamber were covered with aluminum foil to reduce the heat transfer between the two surfaces. The liner was supported on four adjustable legs to minimize heat conduction from the outer chamber wall to the liner. Installation of fitted covers to the liner wall reduced heat transfer through the chamber portholes. During operation the inner wall of the liner normally reached -290 F, while its outer wall read -275 F. The 100-gallon insulated dewars filled with liquid nitrogen supplied the cryogenic system.

The tubular system consisted of the specimen tube, its entrance tube, and the flex hoses used to connect the tubes to the other systems in this experiment. Flexible hoses used to connect the systems were made of stainless steel lined with teflon and had an inside diameter of 0.5 inches. The specimen tube under test was suspended horizontally from the top of the chamber liner on two thin nylon cords very long in comparison to their diameter to minimize conduction losses. A short fitting at the exit end of the tubing permitted a thermocouple measurement of the fluid temperature as it left the instrumented specimen tube. Here fluid flowed through a flexible hose insulated with radiation shielding made of 40 wraps of crinkled 0.001-inch aluminized mylar to the exit port on the chamber. A similar radiation shield was placed around the entrance tube between the chamber door and the front nylon coupling to the specimen tube.

Cajon Ultra-Torr fittings were used to vacuum seal entrance and exit tubes at the chamber flanges on the door and exit port. A thermocouple gland and stainless steel tee were attached to the front of the entrance tube to permit a fluid temperature measurement before the working fluid entered the specimen tube. A metal Swagelok union fitting with nylon ferrules at the front end of the entrance tube facilitated introduction of the plug gage into the tubes for alignment purposes. Fourteen 30-gage copper-constantan thermocouples were used to measure the temperature distribution along the specimen tube. Additionally, one was used to measure the temperature on the inside wall of the cryogenic liner, and another one was used to measure the exit fluid temperature as previously described. These 16 thermocouples were connected to 12-gage thermocouple lead wires with transition junctions where the larger thermocouple lead wire was inserted through the vacuum chamber wall by means of four vacuum feedthroughs. Outside the chamber, thermocouple lead wires were connected from the feedthroughs to an ice-bath reference junction and then to strip-chart recorders located in an adjacent recording room.

Constantan thermocouple lead wires were connected to copper wires, insulated with General Electric RTV silicone sealant at the reference junction, and placed inside an insulated dewar filled with a crushed ice and water mixture. A multipoint strip-chart recorder sampled each of 12 thermocouples for 10 seconds, amplified its signal through one common amplifier, and printed the appropriate thermocouple number at the temperature location on the continuously moving strip chart. These features permitted fast calibration, easy monitoring, and simple data reduction. Eight Bristol strip-chart recorders used in addition to the multipoint to record data such as tube temperatures, water temperature, liner temperature, helium ullage pressure, and water pressure. Strip-chart recorders were calibrated before each experiment with a Leeds and Northrup Type 8690 precision potentiometer.

Consisting of a heating chamber and an electronic temperature controller, the temperature control system was used to control the inlet fluid temperature to 75 $\pm$ 0.5 F. A source of pressurized gas, a pressure regulator, and a water reservoir made up the flow pressurization system. Because of its high insolubility in water, helium gas, stored in cylinders, was used to provide ullage pressure at the top of the reservoir; thus forcing the water through the bottom drain and a 10-micron filter, and into the temperature control system. An open-system arrangement was preferred to a closed system to maintain a constant and known water inlet temperature to the specimen tube within the vacuum chamber. Distilled water was used as the working fluid for this thermal modeling investigation. Eighty-two gallons were stored in a glass-lined water heater reservoir. The large capacity of this reservoir provided an adequate volume for a complete experiment, yet yielded a very slow change in head pressure due to the falling level of the water in the water tank. This slow change in head pressure reduced the need for adjustment of the ullage pressure while awaiting stabilization of a steady-state temperature distribution down the specimen tube. A pressure regulating valve was used to control ullage pressure of a gaseous helium at 20 psig, and a pressure relief valve provided safety. A strain-gage type pressure transducer was used to sense ullage pressure.

To insure that no air was in the entrance and specimen tubes of the tubular system, water was forced to flow down to the entrance tube and up from the specimen tube inside the vacuum chamber. Moreover, a vacuum pump was used to evacuate the tubular system prior to water introduction and then to draw the water through the tubular system and into the fluid collection system. After valving off the flow measuring and collecting systems, this pump pulled a high vacuum on the tubular system at the exit tube and then drew the water from the water tank through the entrance and specimen tubes. A five-gallon, vacuum-transfer safety bottle prevented the introduction of water into the vacuum pump. A valve was used to close off this suction system when smooth and airless flow of water was obtained.

The effectiveness of this air-bleeding operation could be determined by comparing the two sets of thermocouples located the same distance down the specimen tube. One set in the middle of the tube was located at the top and bottom surfaces, while the other set near the exit end of the tube was located on the top and side surfaces. These two sets of thermocouples were used to indicate the peripheral heat flux about the tube and were invaluable in indicating the presence of entrapped air within the specimen tube.

It was important that similar flows be used in each tube so that the effectiveness of the modeling criteria could be studied. The Reynolds number, a dimensionless flow parameter, was selected as the criteria for flow similarity. For a given fluid and tube inside diameter (I.D.), the Reynolds number is related to the volume flow rate in gallons per minute or grams of water per minute. This fact provided a simple means of flow rate calibration. Water was collected in a beaker for one minute and weighed on a set of Ohaus Triple-beam laboratory balance scales accurate to 0.1 gram. Balance was calibrated against a set of certified standard weights. Timing was accomplished with a stopwatch whose accuracy was also certified.

The upstream or head pressure was held at 20.0 ± 0.05 psig by means of a Heise pressure measuring gage and a strain-gage type pressure transducer. The flow was then passed through a 10-inch Brooks rotameter modified to have a range capability of from 0.00003 to 0.04 gallons per minute of water. This was accomplished by using a rotameter that had a very low flow rate capability and shunting the rotameter with a fine micrometer needle valve. It should be pointed out that the calibration curves were used merely as a guide and that an on-line measurement of flow rate was made during each run of the experiments.

This insured an accuracy of flow rate measurement which exceeded the repeatability of the rotameter.

A 16-turn needle valve was used to control the flow and provided a sufficient fine adjustment to this critical parameter. The flow then passed through a tee that provided a choice of two paths, each of which could be shut off with a valve. One path was to the top of the two spherical storage containers of the flow collection system. The other path was to a height identical to that of the former path to the collection system and then to an open tube which permitted collection of the water in the beaker for a flow rate determination. The same height for each flex hose path was important to give the same back pressure during either the flow measurement or collection in the storage spheres.

The flow collection system consisted of two 40-gallon spherical tanks manifolded together to provide adequate storage for the water during an experiment. Water flowed into the top of each sphere rather than the bottom to provide a constant back pressure. Two tubes extended above the tanks provided venting of the displaced air.

Use of a closed collection reservoir system also permitted its pressurization to cycle the water back into the water tank at the conclusion of a day's run. Vent tubes were capped, the valve on the control system was closed, the helium pressurization system was connected to the collection system, and the resulting pressurization of the spherical tanks forced the water through the manifolded bottom of the spheres. From this point, the water flowed through a 40-micron filter and returned to the water tank reservoir.

RESULTS AND DISCUSSION

Experimental test data for the ½" stainless steel and ½" aluminum specimen tubes are shown in Figures 3 and 4. Although fluid inlet temperature to the entrance tube was held to 75 ± 0.5 F, the different lengths of entrance tube along with the different flow rates contributed to a varying degree of heat loss prior to fluid introduction to the specimen tube. Temperature measurement of fluid temperature at the specimen tube entrance was not possible without disturbing the established hydrodynamic boundary layer. Because of this variation it was necessary to normalize the data graphically to a consistent temperature of 65 F at thermocouple number 1 located at $z^* = 0.05$. The selection of this temperature required the least shift of fluid temperature at this location. The associated difference in thermal radiation is negligible. Results of this normalization are presented in Figures 5 through 6 for different Reynolds number and material as a function of tube diameter.

Comparison of the normalized data shows that the same temperature distribution down the tube occurs at Re = 45 for the 1.0-inch O.D. tube, Re = 40 for the 0.75-inch O.D. tube, Re = 35 for the 0.5-inch O.D. tube, and Re = 25 for the 0.25-inch

O.D. tube. Furthermore, the distributions and flow rates correspond well for both stainless steel and aluminum tubes. These results are shown in Figure 7, 8, and 9. The Reynolds numbers mentioned above gave a temperature difference less than 1 F for the stainless steel tubes and less than 2 F for the aluminum tubes. Consistency between both materials for the same Reynolds numbers was within 3 F. These temperature differences are also presented as percent error in terms of absolute temperature and percentage of total temperature difference down the tube in the Table 1 below.

TABLE 1
MODELING ERROR

Tube Combination	Temperature Difference	% Error Abs. Temp.	% Total Temp. Difference
Stainless Steel	1 F	0.196	5.88
Aluminum	2 F	0.392	11.75
S. Steel & Aluminum	3 F	0.588	17.65

The slight dispersion of data for the aluminum tubes is probably due to a larger conduction error resulting from the tube's higher thermal conductivity. Heat conduction is seen to be minimal at $z^* = 0.7$. Conduction error at this location is calculated to be 3.2% for the 1.0-inch O.D. tube and 6.45% for the 0.75-inch tube.

The thermal distribution around the cryogenic liner showed marked differences in temperature between the cylindrical shell and the uncooled end plates. Porthole and porthole covers, which were warmer than the surrounding shell, also contributed to the elevated temperature of the surroundings. The effect of this raised surroundings temperature introduced an error of 2.5% to the 1-inch O.D. tube radiating at 50 F.

Calculation of Grashof number requires the knowledge of temperature difference in the fluid at the top and bottom of the tube. This was not measured because of the disturbing effect of an instrument on the hydrodynamic boundary layer. Temperature differences between top and bottom on the exterior of the tube never exceeded 2 F, and an assumed $\Delta T = 1$ F across the fluid gave Grashof numbers ranging from Gr = 7 in the 0.24-inch O.D. tube at 40 F to Gr = 8850 in the 1.0-inch O.D. tube at 70 F. This range extends from the laminar into a region that is possible to have mixed flow (free and forced convection). Very little investigation has been made in this area, and no work could be found for mixed flow in horizontal tubes. It is difficult, however, to see where free convection can play a major role in the heat transfer within the tube.

Experimental verification of the modeling criteria for Nusselt number, $Nu^* = D^* = L^{*2}$ was not attempted since lack of fluid temperature data prevented the determination of h, the convective heat transfer coefficient. It may be pointed out, however, that $Nu^* = D^*$ requires that $h^* = 1$ for the same fluid.

The thermal entry length was calculated to range from z = 1.6 inches in the 0.25-inch O.D. tube at Re = 25 to z = 12.8

inches in the 1.0-inch O.D. tube at Re = 45.

The relationship between $Re^* = Re_m/Re_p$ and D^* may be determined from the experimental data presented in Figures 7, 8, and 9. Values for these parameters are given in Table 2 below.

TABLE 2
CALCULATION OF Re^* AND D^*

D_o (in.)	D_i (in.)	D_i^*	Re	Re^*
1.0	0.7629	1.0	45	1.0000
0.75	0.5625	0.73732	40	0.88889
0.50	0.3852	0.50492	35	0.77778
0.25	0.1913	0.25075	25	0.55556

These values are presented graphically in Figure 10.

A least-square curve fitting routine for a parabolic distribution was programmed on a computer. The resulting equation was found to be $Re^* = 0.29909 + 1.15314D^* - 0.45605D^{*2}$ over the range from $D^* = 0.25$ to $D^* = 1.0$. This resulting curve is also given in Figure 10. The overall estimated error was $-8.3 \pm 5.3\%$.

CONCLUSIONS AND RECOMMENDATIONS

Fluid flow rate was found to have a large effect upon thermal modeling, and its measurement at very low Reynolds numbers presented difficulties. With regard to Reynolds number criteria, no other investigations have been published in this area, therefore a comparison of results is not possible.

The experiment and its analysis was complex. In order to obtain a sufficient temperature distribution along the tube to allow thermal modeling, the flow rate had to be reduced to a point where the presence of mixed flow was possible. Complete thermal isolation of the specimen tube was impractical due to end connections which permitted some conduction error, particularly in the aluminum tubes. Fluid temperature measurements within the tube could not be made without disturbing the hydrodynamic boundary layer; hence, a complete analysis of convective heat transfer was not possible. The temperature distribution within the cryogenic liner was not close to being isothermal and low. This problem would have been minimized by fabrication of the cryogenic liner from a material with a high thermal conductivity such as brass or aluminum rather than stainless steel.

In spite of these problems it is felt that the investigation provided meaningful results which were previously unavailable and represents an initial inquiry into thermal modeling with three-mode heat transfer including forced convection.

Recommendations for further investigations include the study of mixed flow in a simulated space environment and its effect upon thermal modeling with forced convection at higher flow rates, and further definition of the modeling criteria for the Reynolds number.

The study of mixed flow in a simulated space environment could be accomplished in a setup similar to the one used for this investigation. A thin-walled tube of low thermal conductivity could be thermally instrumented along its upper and lower surfaces and the temperature distributions could be studied for various tube orientations and flow rates.

Thermal modeling with forced convection at higher flow rates may be possible by using higher inlet fluid temperature. Increased radiation heat transfer at higher temperatures may yield a sufficient temperature drop along the tube to permit modeling evaluation. Although tube length in experiment was limited to 48 inches by the chamber dimensions, the use of a coiled tube may provide an effective extended length.

The modeling criteria for Reynolds number should be evaluated under test conditions which preclude the possible effects of mixed flow. Conduction error can also be reduced by use of teflon union fittings or similar devices for connection of the specimen tube to the adjacent plumbing.

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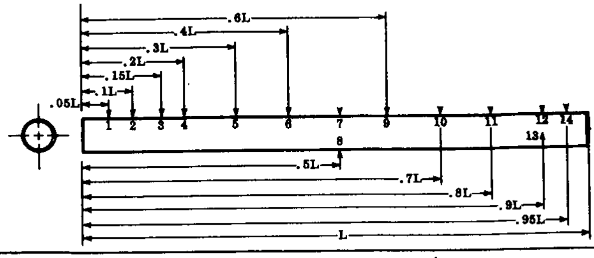
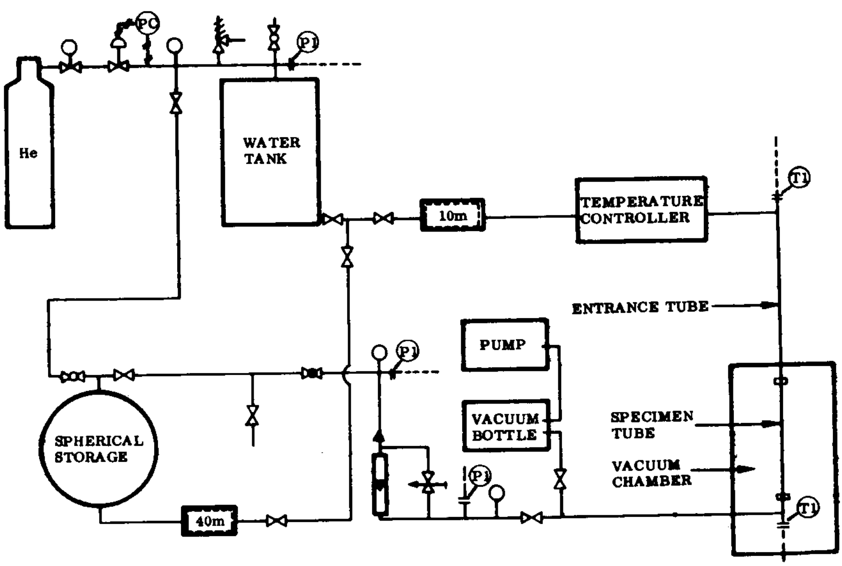


	TUBE LENGTH DIMENSION*												
	L	.05L	.10L	.15L	.2L	.3L	.4L	.5L	.6L	.7L	.8L	.9L	.95L
Thermocouple Number		1	2	3	4	5	6	7 & 8	9	10	11	12 & 13	14
Tube Diameter													
1.0	48.000	2.40	4.80	7.20	9.60	14.40	19.20	24.0	28.8	33.6	38.4	43.2	45.6
.75	41.568	2.078	4.157	6.235	8.314	12.470	16.627	20.784	24.941	29.098	33.254	37.411	39.490
.50	33.936	1.697	3.394	5.090	6.787	10.181.	13.574	16.968	20.362	23.755	27.149	30.542	32.239
.25	24.000	1.20	2.40	3.60	4.80	7.20	9.60	12.00	14.40	16.80	19.20	21.60	22.80

* All dimensions are shown in inches.

Figure 1. THERMOCOUPLE LAYOUT ON SPECIMEN

Figure 2. PRESSURIZATION AND FLOW SYSTEMS SCHEMATIC DIAGRAM



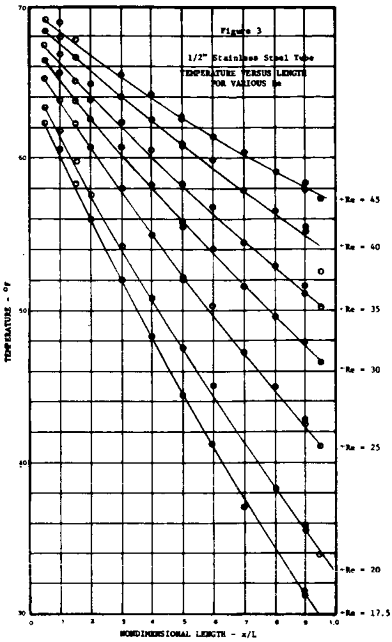


Figure 3
1/2" Stainless Steel Tube
TEMPERATURE VERSUS LENGTH
FOR VARIOUS Re
Re = 45
Re = 40
Re = 35
Re = 30
Re = 25
Re = 20
Re = 17.5
TEMPERATURE - °F
NONDIMENSIONAL LENGTH - x/L

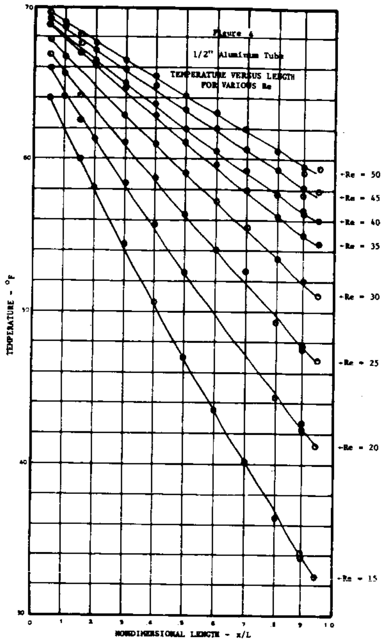


Figure 4
1/2" Aluminum Tube
TEMPERATURE VERSUS LENGTH
FOR VARIOUS Re
Re = 50
Re = 45
Re = 40
Re = 35
Re = 30
Re = 25
Re = 20
Re = 15
TEMPERATURE - °F
NONDIMENSIONAL LENGTH - x/L

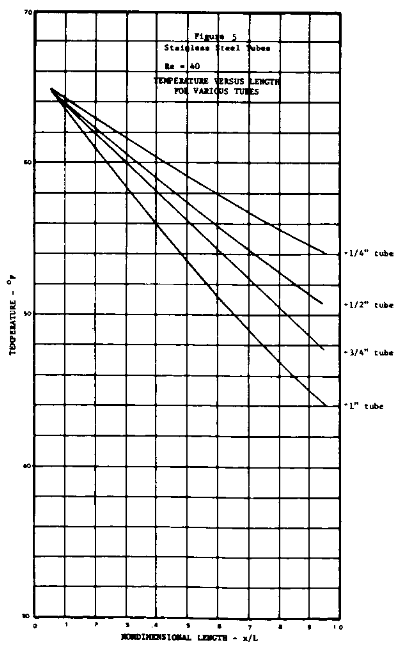


Figure 5
Stainless Steel Tubes
Re = 40
TEMPERATURE VERSUS LENGTH
FOR VARIOUS TUBES
1/4" tube
1/2" tube
3/4" tube
1" tube
TEMPERATURE - °F
NONDIMENSIONAL LENGTH - x/L

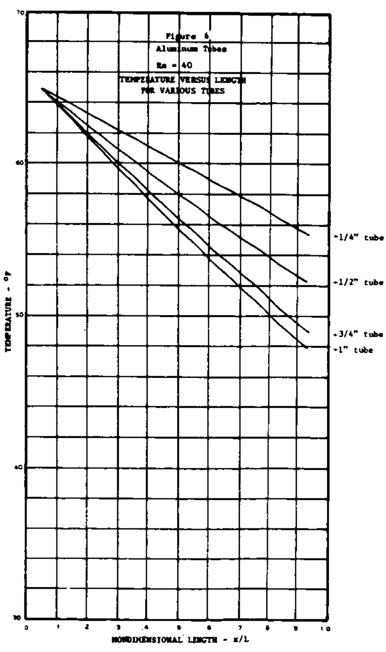
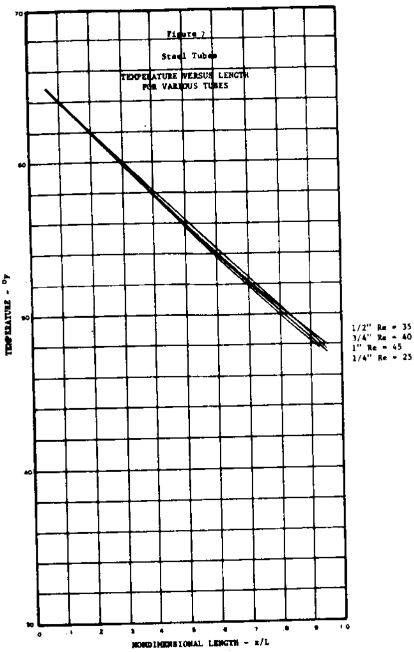
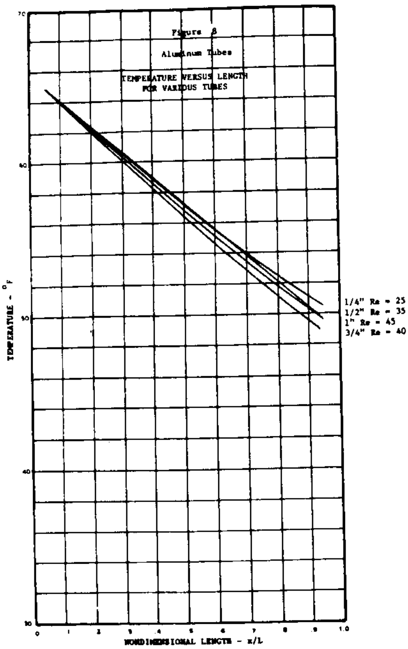
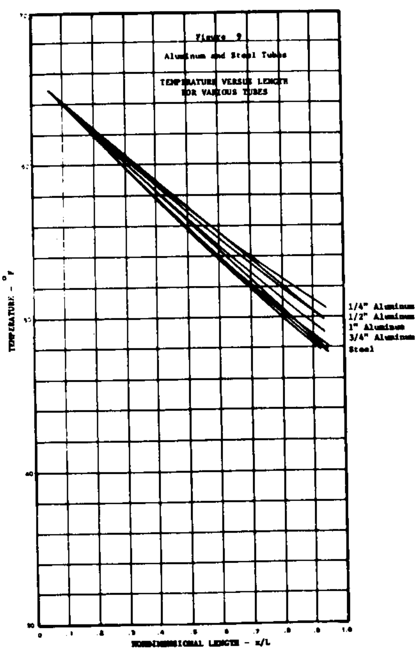


Figure 6
Aluminum Tubes
Re = 40
TEMPERATURE VERSUS LENGTH
FOR VARIOUS TUBES
1/4" tube
1/2" tube
3/4" tube
1" tube
TEMPERATURE - °F
NONDIMENSIONAL LENGTH - x/L







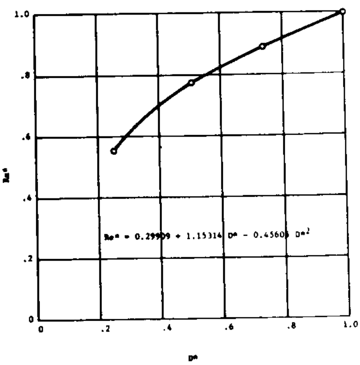


Figure 10. Graphical Relationship Between Re* and D*